ROTORDYNAMIC ASPECTS OF A NEW HERMETICALLY SEALED PIPELINE COMPRESSOR

by

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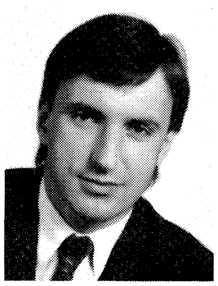

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ABSTRACT

Mopico is an oilfree motor pipeline compressor of fully integrated design with no sealing requirements against atmospheric pressure and supported on magnetic bearings. The application of magnetic bearings for this machine represents todays extreme combination of weight and speed. The rotor weighs one ton and runs at a maximum speed of 10000 rpm. Hence, it is not surprising that a variety of rotordynamic problems had to be solved before the machine could be operated safely. The magnetic bearings are described and the experience gained during the development of the machine is reported, based on the experimental and theoretical investigations.

The important properties of magnetic bearings are the capacity limits, the damping and stiffness of the bearing and destabilizing effects, which are of a different nature to those known for compressors supported on conventional oil bearings.

In the failure case, the rotor is dropped into auxiliary bearings, which are ball bearings with clearance between rotor and the inner race. The most severe condition in this emergency case is a whirling motion of the rotor, with the clearance as the radius of the orbit. The load generated on the bearings due to the centrifugal forces of this motion are very high and can destroy the bearing, if it lasts for an extended period. A corrugated ribbon between the outer race of the bearing and the housing has a damping effect in the circumferential direction and stops the whirling after a short time.

INTRODUCTION

It is an old dream of many turbomachinery manufacturers to build a hermetically sealed compressor since, in this case, no sealing against atmospheric pressure is necessary. However such a design requires that the whole rotor, including drive and bearings, is contained within the compressor housing. This turbodream has now been realized for the first time in a motor pipeline compressor (MOPICO). Two recent developments enabled its realization in a rather simple way: high speed electrical motor drives and magnetic bearings.

A cross section of the compressor is shown in Figure 1. The whole rotor is surrounded by gas and the variable speed motor, which is in the middle of the rotor, is cooled directly by the compressed gas. The compressor has two impeller stages mounted at the shaft ends. The stages can be operated in parallel or series modes. The machine combines unique features such as no emissions, remote controllability, broadest operating range at high efficiency, along with low maintenance and installation costs.

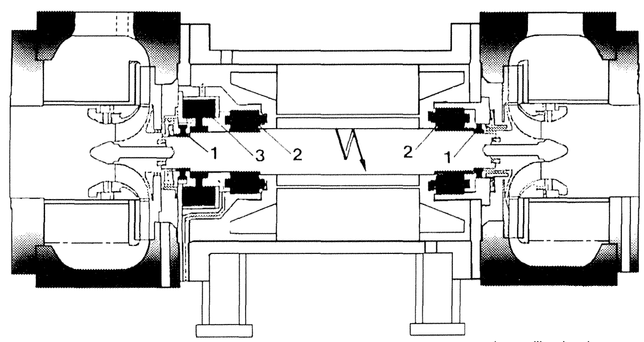

Figure 1. Cross Section of the Motor Pipeline Compressor.

The electrical motor of the MOPICO has set a new record concerning speed and power. It is the first time that a six megawatt electrical motor has run at a speed as high as 10000 rpm.

According to the S2M reference list of January 90 (S2M is the worlds leading magnetic bearing manufacturer) the application of magnetic bearings for a rotor of the weight of one ton and a speed of 10000 rpm is also unique (Figure 2).

The rotordynamic aspects of the compressor are reported. The magnetic bearings, their properties and the vibrational behavior of the rotor in magnetic bearings are described, along with the rotor being dropped into the auxiliary bearings after failure of the magnetic bearings. Vibrations in axial direction are not considered, since they are not critical.

DESCRIPTION OF THE MAGNETIC BEARINGS

The principle and the elements of a magnetic bearing are shown schematically in Figure 3. The displacement of the rotor is measured by a sensor. The sensor output signal (voltage) is modified

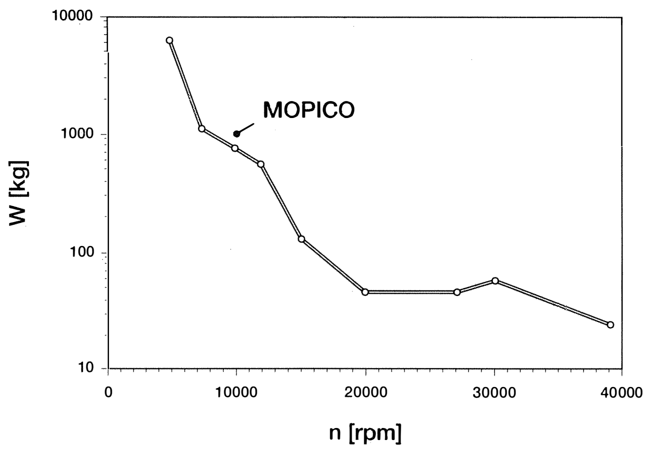

Figure 2. Extreme Speeds and Weights of Large Turbomachines Supported on Magnetic Bearings (S2M reference list).

in the controller and amplified. The amplifier output is a current proportional to its input voltage. The bearing force is generated by the current in the actuator, which is an electromagnet. One radial bearing consists of two such loops for two independent radial directions. The force of the magnetic bearing has to support the rotor weight and damp rotor vibrations, hence, it must have stiffness and damping properties, which are described in the following.

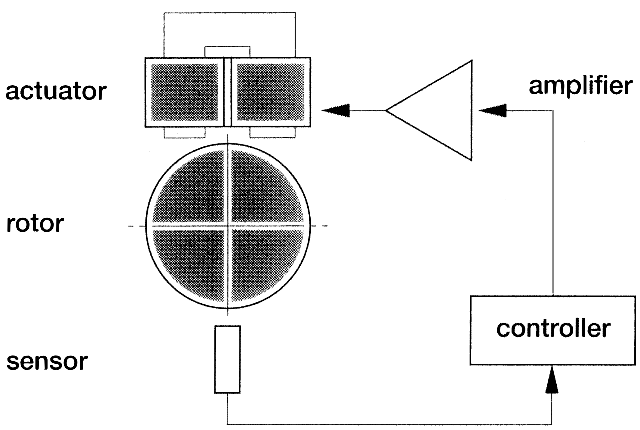

Figure 3. Schematic Drawing of a Magnetic Bearing.

Stiffness and Damping Properties of the Magnetic Bearings

The relation between rotor displacement and bearing force may be qualitatively considered as the relation between input and output of the controller, since the other elements (sensor, amplifier, and actuator) basically produce output signals proportional to their input signals. Hence, the stiffness and damping properties are mainly determined by the controller (apart from a simple factor due to the combined influence of the other elements). In the following, the controller and its effect on the stiffness and damping of the bearing are explained. This requires a short excursion into control theory and thinking in the frequency domain.

Magnetic bearings normally have PID controllers. This means that the output signal of the controller is the sum of the following three parts:

- a signal proportional to the input signal, that is the rotor displacement (the P-part),
- a signal proportional to the integral (the I-part), and
- to the derivative of the input signal, that is the velocity of the rotor (the D-part).

The three parts are illustrated in Figure 4 for a sinusoidal time signal of the rotor displacement x. The P-part may be considered as a spring and the D-part as a damper. The P-part is in phase with

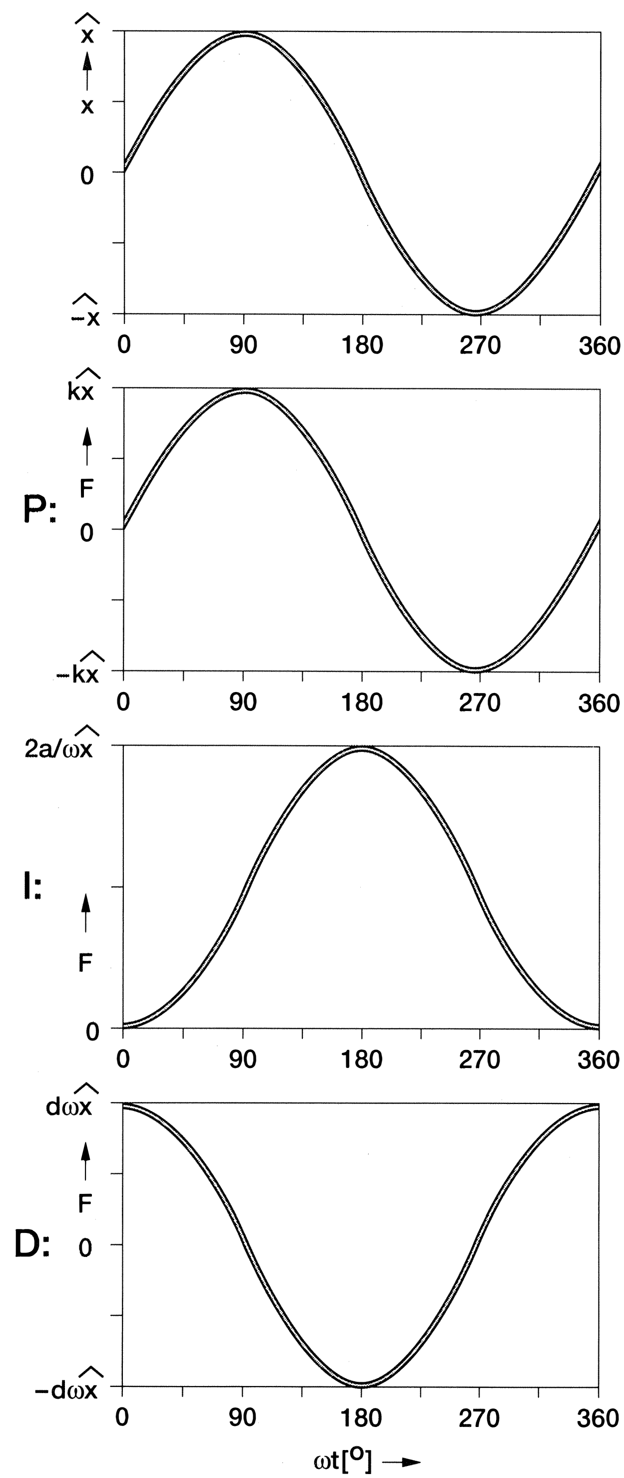

Figure 4. The P-, I- and D-part of a Bearing Force for a Sinusoidal Displacement X.

the rotor displacement, whereas the D-part is 90 degrees ahead of the input signal (it has its first maximum already at wt = 0 degrees whereas the sinus has its first maximum only at wt = 90 degrees). Generally it can be stated, that a bearing provides damping, if the force has a phase lead relative to the rotor displacement. The I-part is 90 degrees behind the input signal (it has its maximum only at wt = 180 degrees), hence it has a destabilizing effect. In the following is explained, why the I-part is nevertheless useful, although the P- and D-part alone can create stiffness and damping as a conventional oil bearing.

The amplitudes of the different parts of the PID controlled bearing force are shown in Figure 5 in relation to the amplitude of a sinusoidal rotor displacement as a function of the frequency of the sinus. It can be seen that the I-part provides high force amplitudes at low frequency and decreases rapidly at higher frequencies. For the frequency zero, that is for a static displacement, it theoretically provides an infinite force, hence, theoretically, the I-part prevents any static deflection of the rotor in the bearing. Practically it is very small. This is very important for magnetic bearings, since their force characteristic is very sensitive to large deflections. This is due to the nature of the magnetic force, which increases as the gap between rotor and stator decreases and the current in the actuator remains unchanged. Hence, the gap dependent part of the magnetic field force acts against the current dependent part. This is also the reason, why a magnetic bearing could not work without a controlled current. The rotor would always "stick" to the stator.

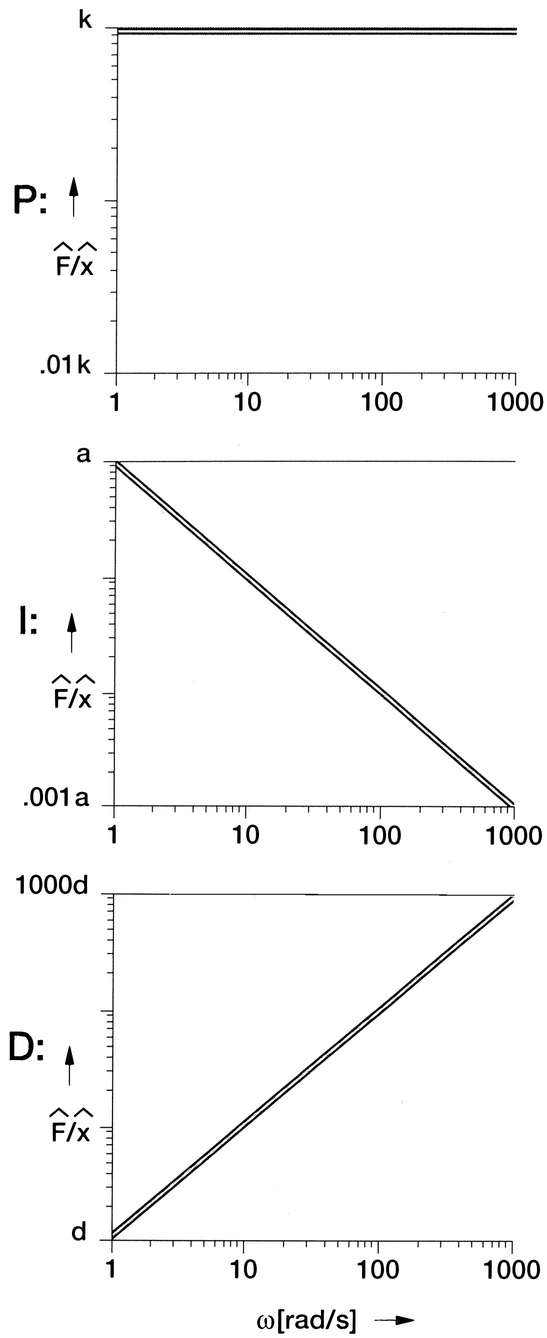

Figure 5. Relative Amplitudes of the P-, I- and D-part of a Bearing Force for Different Frequencies of a Sinusoidal Displacement.

The high forces at low frequencies due to the I-part allow a low stiffness of the bearing, that is a small P-part, without causing large static deflections. At frequencies, where the I-part is no longer dominating, the vibration amplitudes in the bearings are relatively high, compared to the amplitudes at other locations within the rotor, and the bearing damping force is more effective in reducing the vibration level. This effect is also present in conventional oil bearings, where bearings with a large clearance have a better damping effect on the rotor. However, in this case, aerodynamic forces sometimes cause trouble. These forces have low frequencies, hence act rather like static forces and cause high rotor deflections due to the low oil bearing stiffness for large clearances. In the case of magnetic bearings this can be prevented by the I-part.

Superimposing the amplitudes in Figure 5 yields the amplitude of the transfer function between bearing force and displacement, as it is quantitatively shown in Figure 6 for the MOPICO bearings. The phase angle of the transfer function in this figure is the angle between the superimposed force and the displacement (as in Figure 4 for the single P-, I-, D-parts) for different frequencies. The transfer function in Figure 6 is not that of a bearing with an ideal PID controller as in the Figures 4 and 5, since such a controller is not realized by the electronics in the analog controller. However, it comes very close to an ideal PID transfer function.

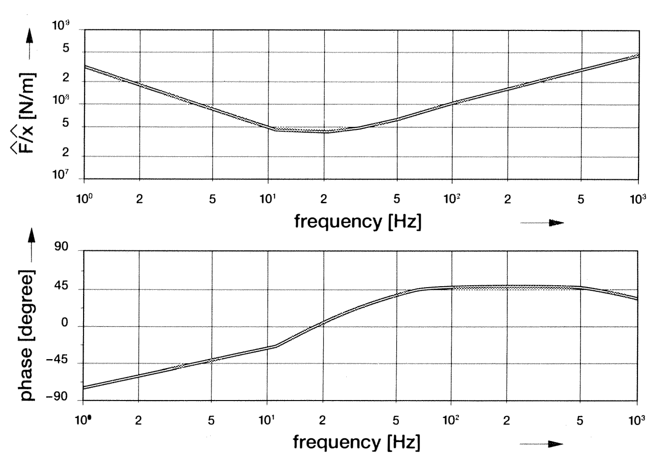

Figure 6. Transfer Function Between Force and Displacement of the Magnetic Bearings.

At low frequencies, the influence of the I-part can be clearly recognized by the high amplitude and the negative phase angle. For higher frequencies, the D-part dominates, hence the amplitude increases with increasing frequency and the phase angle is positive.

From the transfer function in Figure 6, equivalent damping and stiffness coefficients can be calculated by determining for each frequency such coefficients (Figure 7), which yield the same relative force amplitude and phase angle. They depend on the vibration frequency of the rotor; however, they do not depend on the rotating speed. This is new compared to oil bearings, which depend on the rotating speed but not on the vibration frequency.

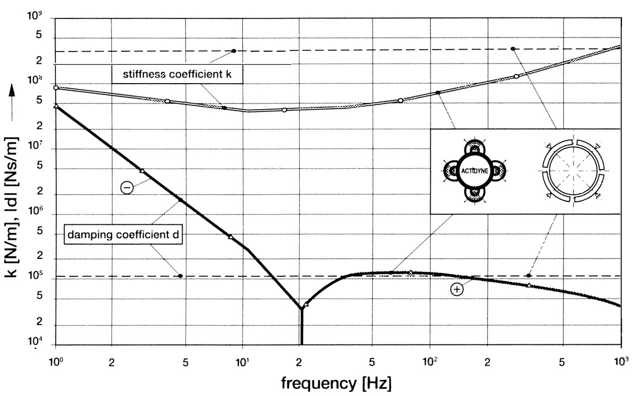

Figure 7. Damping and Stiffness Coefficients of the Motor Pipeline Compressor Magnetic Bearings and Oil Bearings.

What is the difference, since a rotor always vibrates with the rotating speed due to unbalance? For the unbalance vibration there is indeed no difference. However, a rotor also vibrates with other frequencies, mainly natural frequencies due to transient excitations from aerodynamic or other forces.

For comparison, the damping and stiffness coefficients are also shown in Figure 7 of a four tilting pad oil bearing for the maximum speed of 10000 rpm (pad inertia neglected), which would suit this compressor. It can be seen that the magnetic bearing stiffness is much lower than that of the oil bearing. In a frequency range from 10 to 200Hz, it is about one third to one sixth of the oil bearing stiffness. The damping coefficient of the magnetic bearing is negative at low frequencies (below 20 Hz). This is due to the I-part in the controller, which causes a phase loss between force and displacement, whereas a positive damping causes a phase lead, as is explained above (Figure 4). In the frequency range from 50 to 200 Hz the damping of the magnetic bearing has the same order of magnitude as that of an oil bearing.

Destabilizing Effects of Magnetic Bearings

The rotor bearing system can be destabilized by the magnetic bearing if any natural vibration (those vibrations, which are excited by transient forces) of the combined system is excited rather than damped by the bearing force.

In the previous paragraph, it was shown that at low frequencies, the damping of the magnetic bearings is negative, due to the I-part in the controller. Apart from this low frequency range, the damping may also become negative at very high frequencies above the range shown in Figure 7, due to phase losses of the force relative to the displacement in the other elements (sensor, amplifier, and actuator), which are not taken into account in the previous paragraph, since they are not apparent in the frequency range considered there.

Besides the negative damping, the rotor can also be excited from the bearing, because the sensor and actuator are not in the same axial position. An example of a natural vibration mode with a node lying between sensor and actuator is shown in Figure 8. In such a situation, the sensor gives a false signal to the actuator, since the displacement at the actuator is in the opposite direction to that at the sensor.

The excitation of the rotor by the bearing at very high frequencies normally does not cause any trouble, since other damping forces like the internal damping are very effective in this frequency range.

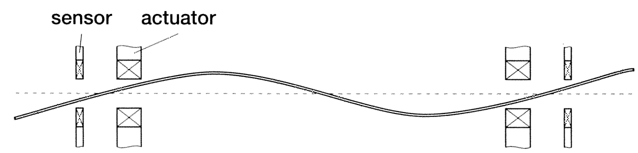

Figure 8. Example of a Mode Excited by the Magnetic Bearing.

The excitation due to negative damping in the low frequency range (below 20Hz in case of this compressor) also does not do any harm, as long as the lowest natural vibration mode has a frequency above this range. This can easily be achieved by an appropriate bearing stiffness.

However, in the medium frequency range (above 20Hz and below approximately 1000 Hz in the case of the motor pipeline compressor), one has to be careful. The damping must not be negative (which is the case for this compressor) and the location of the bearing has to be chosen very carefully, in order to prevent a node of the natural vibration modes occurring between sensor and actuator or even close to them, since the best damping is provided in case of large displacements in the bearing relative to other locations. The latter requirement is very difficult to fulfil for every natural vibration mode in this frequency range, since there are several modes, all of which have different nodes and one is always restricted in positioning the bearing due to other design requirements. However, it must be fulfilled for all modes below the maximum speed in order to damp all critical speeds sufficiently. If other modes beyond the maximum speed are excited by the bearing, the bearing force has to be filtered at the frequencies of these modes.

The proper tuning of magnetic bearings (changing the controller to achieve the proper damping and placing the filters) is a quite time consuming job (two to four weeks), since the controller is analog, and tuning means exchanging electronic components. Digital controllers promise considerable progress in this regard, because changing the controller then can be done by changing the program of a signal processor. Magnetic bearings with digital controllers are already realized and will probably soon be applied for compressors also.

Capacity Limits of the Magnetic Bearings

The capacity limits of magnetic bearings are mainly determined by the amplifier and actuator. The actuator of a magnetic bearing is shown in Figure 9 with its poles and the path of the magnetic fields. The windings in the stator (not shown in the figure for reasons of clarity) and the current in the windings determine the magnetic fields, hence the poles on the stator. The poles on the rotor adjust themselves continuously according to the stator poles (a north pole opposite a south pole and vice versa) causing attracting forces. In order to achieve this effect, the rotor must have a sleeve of magnetic material. As it rotates, the poles move in the circumferential direction relative to a fixed point on the rotor. This causes eddy current losses. To reduce them, the sleeve on the rotor is laminated. The whole bearing consists of four quadrants, two opposite quadrants being coupled to create the bearing force in one direction. Each of the two independent directions needs two quadrants, since the bearing must be able to create forces with a plus and minus sign, and each quadrant is only able to pull. The two independent directions of the bearing have an angle of 45 degrees to the vertical direction, so the bearing forces of both directions contribute to carry the rotor weight.

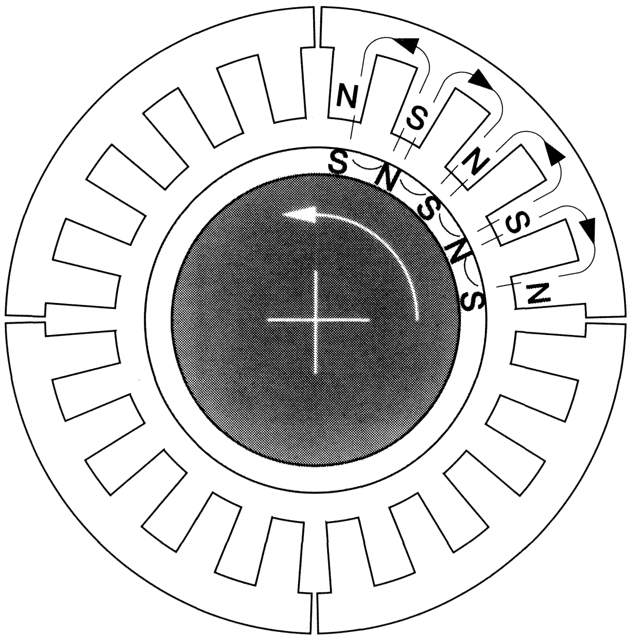

Figure 9. Actuator of the Magnetic Bearings.

The force current relation (values for the motor pipeline compressor bearings) for one quadrant is shown in Figure 10. For small currents, the relation is a square function. For higher currents, the saturation of the magnetic bearing becomes effective, and the increase in force with increasing current becomes smaller. It is obvious that the relation between current and force is not linear as assumed in the the previous sections and as is necessary to apply the well developed control theory. To achieve a linear relation in a certain current range, a bias current is applied, which has a value

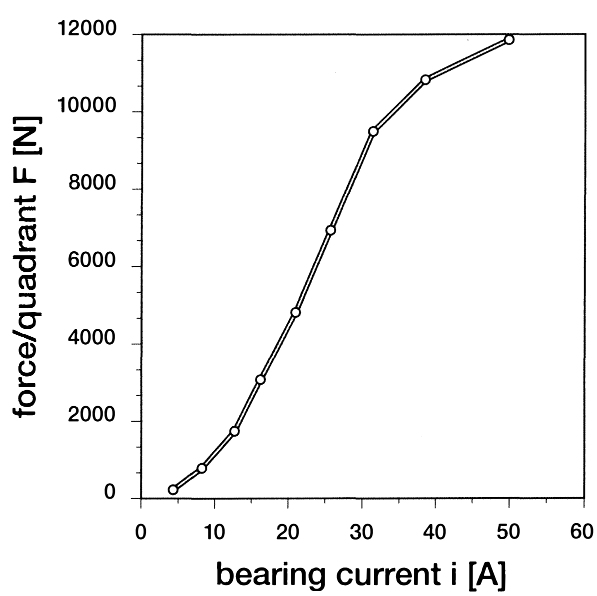

Figure 10. Force-Current Relationship for One Quadrant.

approximately in the middle between zero and saturation. If no force is required in one direction, both quadrants for this direction pull with an opposing force corresponding to the bias current. If a force is required, the current is increased in one quadrant and decreased in the other. The maximum force is created when the current in one quadrant is zero and twice the bias current in the opposite quadrant. This maximum force is determined by the magnetic saturation and depends on the magnetic material which is used. Specific forces of 800 bar can be achieved. These values are much smaller than those of oil bearings, where specific loads of 2500 bar can be realized.

Other force limiting factors are caused by the amplifier. The amplifier can only provide a limited current. The maximum current should be in agreement with the maximum force at the magnetic saturation (Figure 10). The amplifier also has a limited voltage. Its maximum voltage is the power supply voltage. Since the electromagnet has a high inductance and the change in current in such a case is proportional to the voltage, the change in current and, thus, in force is limited. This limits the force at high frequencies, since high frequencies mean rapid changes. The resultant force limits of the magnetic bearings used in the compressor caused by the amplifier and the actuator are shown in Figure 11.

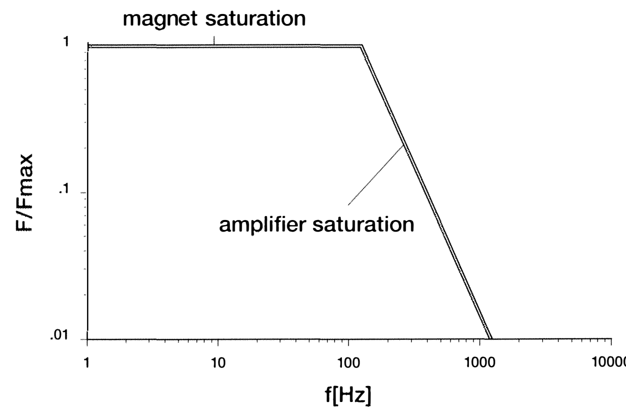

Figure 11. Force Limits of the Magnetic Bearings, Fmax = 10000 N.

VIBRATIONAL BEHAVIOR OF THE ROTOR IN MAGNETIC BEARINGS

The calculated frequencies and damping factors (to obtain logarithmic decrement multiply these by 2π) of the first three (forward whirling) modes of the compressor are shown in Figure 12. The method of modelling the magnetic bearings for the calculation is described in detail in an earlier article [1]. For comparison, the frequencies and damping factors of the rotor supported on suitable oil bearings (the same as those, with damping and stiffness coefficients are shown in Figure 7) are also shown. It can be seen that the rotor on magnetic bearings is much better damped, although the damping coefficient of the magnetic bearing is not higher than that of the oil bearing (see Figure 7). This is due to the low bearing stiffness, which makes the damping more effective. Because of the low stiffness the first two modes of the rotor on magnetic bearings are lower than those of the rotor on oil bearings; hence, there is no critical speed in the operating speed range.

The vibration level of the machine in the final tests was well below the bearing capacity limits in Figure 11. Two measurements are shown in Figure 13 of the once per revolution bearing force amplitude of the rotor for a very fast run up (within two minutes) of the cold rotor to 10000 rpm. Although this is a quite severe, test the margin relative to the limit at the maximum speed is still about

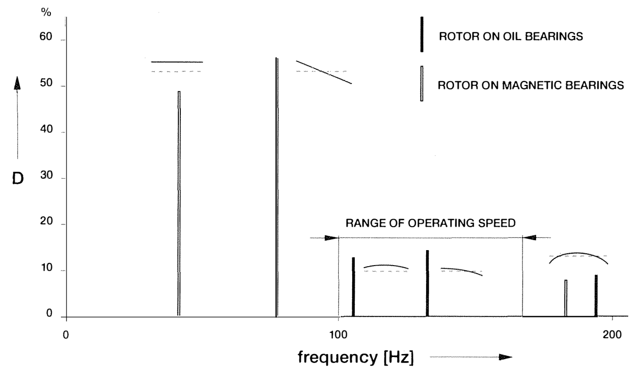

Figure 12. Natural Frequencies and Damping Factors of the First Three Modes of the Compressor Supported on Magnetic Bearings and Oil Bearings.

50 percent of the limit force. The vibration level of the warm rotor is only half the level of that shown in Figure 13.

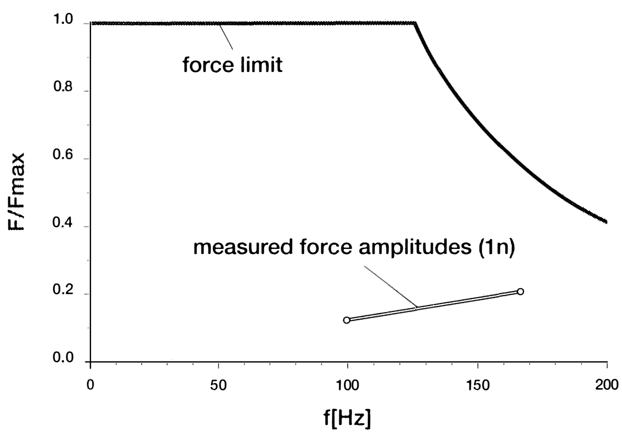

Figure 13. Measured Once per Revolution Force Amplitude Relative to the Bearing Limits.

THE FAILURE CASE

At standstill, and in the case of a failure of the magnetic bearings, the rotor is supported on the auxiliary bearings. They are ball bearings with a clearance between rotor and inner race (Figure 14). The clearance is approximately half the clearance between rotor and stator of the magnetic bearing. For this compressor, it is 250 mm (approximately 10 mils) in radius. The corrugated ribbon between outer race and compressor housing is installed to generate a damping effect.

The dropping of the rotor into the auxiliary bearings was calculated and tested at almost full speed and almost full load. In the test, the rotor was dropped by switching off the magnetic bearing amplifiers. The electric drive was shut down simultaneously. Under these conditions, the aerodynamic torque at the impellers decelerates the rotor very fast following the shutdown.

The calculated orbit of the rotor at the auxiliary bearings is shown in Figure 15. The method of calculation is described in detail in [2, 3]. As the rotor touches the inner race (which is not rotating at the instant of the dropping), the frictional force between rotor and inner race accelerates the rotor in the reverse direction to the rotation and the inner race in the direction of rotation, as illustrated in Figure 16. The frictional force acts on the rotor as long as it is in contact with the inner race and the inner race does not yet have the speed of the rotor. Due to this acceleration, the rotor starts whirling and is pressed against the inner race by the centrifugal force, which is very high and exceeds the load due to the rotor weight by a factor of six and even exceeds the impulse force of the first touch down. It also prevents a bouncing of the rotor, such as can be observed for drops at standstill. If this high load acts on the bearing for an extended period, the bearing may be destroyed, due to the heat generated mainly from the rolling friction, and only to a small extent from the sliding friction between inner race and rotor, since the latter is only present for a short time, until the inner race has the same speed as the rotor.

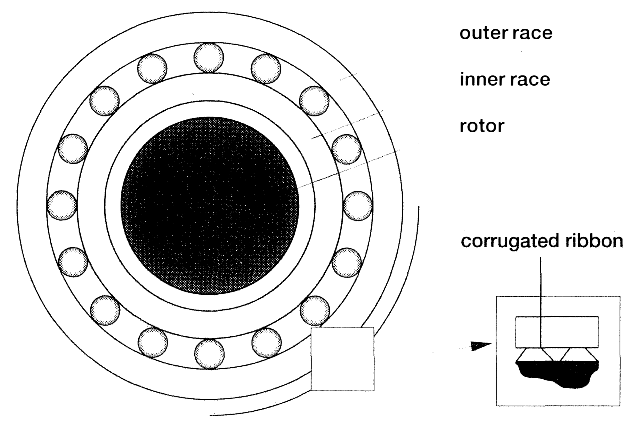

Figure 14. Schematic Depiction of the Auxiliary Bearings.

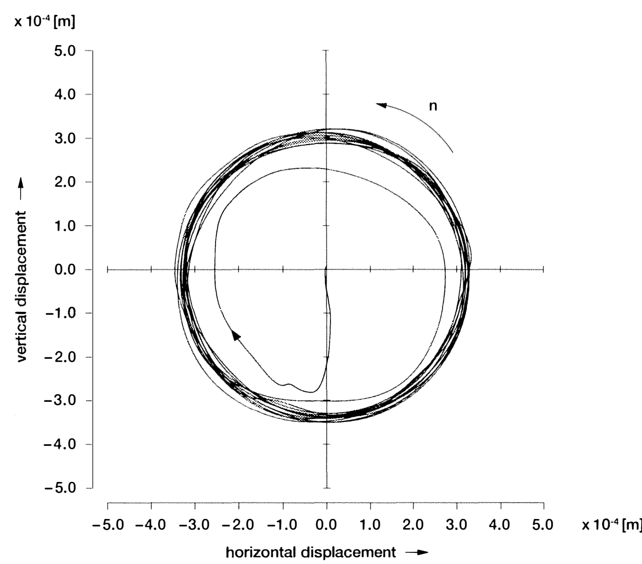

Figure 15. Calculated Orbit of the Rotor Being Dropped into the Auxiliary Bearings.

The centrifugal force depends mainly on the whirling frequency (which has a value of approximately 60Hz in Figure 15) and the radius of the orbit. The radius, however, is mainly determined by the auxiliary bearing clearance, apart from a small elastic deformation component. Hence, the load can primarily be influenced by measures which reduce the frequency. As further simulations showed, the friction coefficient between rotor and inner race, along with as the moment of inertia of the inner race, have the

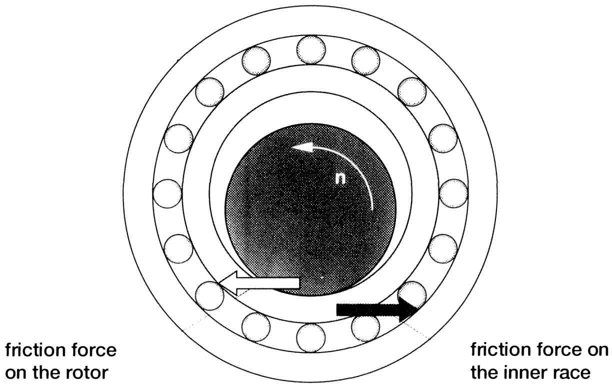

Figure 16. Friction Between Inner Race and Rotor.

largest influence. Reducing both decreases the whirling frequency. If the friction coefficient is below 0.35, whirling does not occur at all. However, such a value cannot be guaranteed, since there is no lubrication. The most effective measure to prevent damage is probably to apply cicumferential damping, to stop the whirling motion after a short time. The corrugated ribbon between outer race and housing has such an effect. (Its circumferential damping was not taken into account in the simulation in Figure 15, only its radial damping.)

The measured orbit is depicted in Figure 17. The rotor also starts whirling but stops after seven cycles. The whirling frequency is approximately 37 Hz. The maximum deflection is much larger than in the calculation. This is because the auxiliary bearing was mounted slightly eccentric when the test was carried out. However, what is completely surprising is the whirling direction. It is not in the reverse direction to the rotation of the shaft as was expected from the simulation. The rotor whirls in the same direction as the shaft rotates.

Possible forces to compel the rotor to whirl in forward direction are labyrinth seal forces and electromagnetic forces in the motor. The labyrinths however can be ruled out, as a calculation including the labyrinths showed.

Electromagnetic forces will still be present after the rotor has dropped, due to the collapsing electromagnetic fields, although the motor was shut down simultaneously. One has to consider also, that the time from switch off to the end of the whirling is only a few tenths of a second, and that the manual motor shutdown could have been slightly later.

It is known from Fruechtenicht, et al. [4], that lateral vibrations in induction motors with a squirrel cage may cause forces in cicumferential direction. These forces can be calculated for the nominal conditions of the motor. They are not large enough to explain the forward whirling, as a calculation showed. The forces may become much larger in the case when the stator field collapses. An accurate calculation for this case, however, is not possible. Further tests will be carried out to clarify the cause of the forward whirling. They were not possible until now, since the machine had to be removed from the test stand right after the test, before the results could be evaluated.

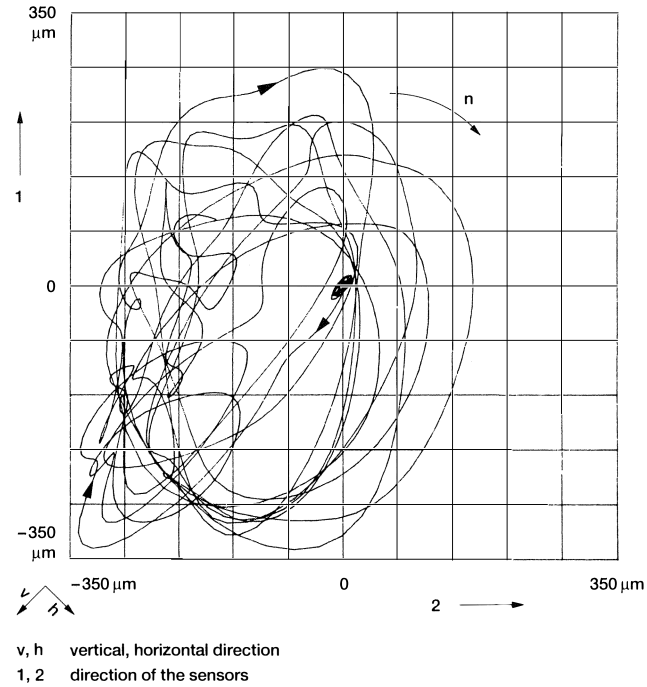

Figure 17. Measured Orbit of the Rotor Being Dropped into the Auxiliary Bearings.

CONCLUSIONS

Without magnetic bearings the realization of hermetically sealed machines is not possible in such a simple and elegant way as in case of the motor pipeline compressor. However, their reputation of being able to solve any rotordynamic problem is not quite true. On the contrary, they can cause a variety of new problems, and one has to be aware of them to apply them properly, and to be able to make full use of their well known advantages (no wear, no lubrication, low power consumption).

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